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M. W. Garland
Frick Company

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LARGE RECIPROCATING COMPRESSOR DESIGN GUIDE LINES

Milton W. Garland, Senior Consultant,
Frick Company, Waynesboro, PA

For the purpose of this discussion, a large reciprocating compressor is assumed to be one of 500 CFM displacement. There are in service reciprocating compressors of 1000 CFM and over, but close to the 500 CFM size there are elements of design which make it advisable to carefully examine the two basic structures of vertical enclosed single-acting compressors now in use.

The design will be the vertical enclosed single-acting type.

The two basic structures to be evaluated are illustrated in Figs. 1 and 2.

Table 1 has been prepared for the purpose of illustrating and comparing some of the "Working Rates" of presently used vertical enclosed type compressors. Note that the term "vertical" is not limited to a compressor with true vertical cylinders only, but to any V and W configuration wherein all cylinders are above the horizontal plane of the crankshaft. All of the compressors listed, except one, are of the Fig. 2 structure. The one of the Fig. 1 structure is the only one unlimited as to type of refrigerant and applicable to a 13 to 1 compression ratio. In the Fig. 2 structure some of the compressors listed are limited to halocarbon refrigerant service only at a maximum 9 to 1 compression ratio, and others that are offered for use with ammonia as well as the halocarbon refrigerants have limiting compression ratios varying from 6.5 to 1 to 8.5 to 1.

This study will follow the theme that the configuration selected will be that most suitable for use with any of the presently used refrigerants such as R-717, R-12, R-22, R-502, R-600, R-600a and R-1270 at condensing pressures obtainable with the air and water temperatures which might be encountered in any part of the world where mechanical refrigeration must be used.

The first step is to make a tabulation of possible compressor configurations relative to RPM, number of cylinders, bore and stroke, etc.

Table 2 has been provided for the purpose of comparing difference in RPM, number of cylinders, and evaluation of the relative merits of the structure of Figs. 1 & 2.

Each item (b) is the 500 CFM divided by the RPM.

In Table 1, column 12, it is recorded that the largest cu. ft. per rev. per cyl. is .0170; therefore

$$\frac{.2777}{.0170} = 16+ \text{ cyls.}$$

It would seem possible to use a 16 cyl. structure and an examination of same will be made. The selection of 16 is recorded under column 1, item (b 1).

A selection of piston speed must be made and the data from Table 1 will be used as a guide. Column 8 of this Table shows a piston speed of 825 ft. per minute average for the two 1800 RPM compressors listed. It is to be observed that the lower RPM compressors have a lower average piston speed.

There is good foundation for this pattern because of the fact that volume increases by the square, relative to the cylinder bore; whereas, the circumference increases directly with an increase in bore. Therefore, when the bore is increased, the circumference for valve seat does not increase proportionately. The suggestion that increased valve lift will provide the solution is not valid because valve performance studies have proven that valve lifts must generally be decreased as RPM is increased.

The following shows the relative change of volume and circumference for a few bore sizes:

Bore Inches	Area Sq. Inches	Circum. Inches
2	3.1416	6.2832
3	7.0686	9.4248
4	12.5660	12.5660
5	19.6350	15.7080
6	28.2740	18.8500

The speed of 825 RPM is recorded under column 1, item (d) and the stroke of 2.75 inches is recorded under column 1, item (f).

The volume per cylinder per revolution is determined by:

$$\frac{.27777 \text{ CFM} \times 1728}{16 \text{ cyls.}} = 30 \text{ cu. in. per cyl. per rev.}$$

The necessary bore is:

$$\frac{30 \text{ cu. in.}}{2.75 \text{ in. stroke}} = 10.909 \text{ sq. in.}$$

$$R = \sqrt{\frac{10.909}{3.1416}} = 1.8609 \quad \text{Dia.} = 3.7218 \text{ inches}$$

The bore size is recorded under column 1, item (e).

The bore to stroke ratio, $\frac{3.7218}{2.75} = 1.354$ and is recorded under column 1, item (g).

It is now necessary to generate data from which valve application and performance comparisons may be made. Item (h) of Table 2 is obtained by simple computation and recorded for future use. The same is true for item (i). The rate to fill a cylinder in cu. in. per second is obtained by:

$$\frac{1 \text{ sec.}}{.01666 \text{ sec.}} \times 30 \text{ cu. in.} = 1800 \text{ cu. in. per sec.}$$

and this is recorded under column 1, item (i).

As many other combinations of speed, number of cylinders, etc., which might be of interest and value should be recorded in the manner described above.

For this discussion, we will add under column 2 of Table 2 the speed of 1200 RPM using 16 cyls. and a reduction of piston speed to 720 ft. per min. which results in a much more favorable Bore-to-Stroke ratio.

Under column 3 of Table 2 there is recorded another 1200 RPM possibility, but with 8 cylinders. This choice was made because of the fact that a 16 cyl. structure involves a long shaft with center bearing, manufacturing difficulties, and an out of proportion cost. The 8 cyl. structure presents the possibility of eliminating the center bearing and two shaft throws.

Under column 4 of Table 2 there is recorded a 900 RPM possibility with 8 cyls.

All of the items, (a) to (i) inclusive, are recorded at this time.

Structural considerations are next in order and item (j) is selected as .008 inches for all four compressors. It is to be noted that in Table 1, column 9, the estimated distances are the design minimums. The .008 inch selection is an estimated average of minimum to maximum manufacturing tolerances.

Item (k1) is item (j) multiplied by the bore area.

The various types of valve structures which might be incorporated into the design must now be evaluated and recorded. Fig. 4 is representative of the type of valve characteristics which the designer should prepare for the purpose of determining the most acceptable performance in the final compressor structure. Those illustrated are typical but do not represent all of the sizes or configurations currently available or in use. One of the objectives is to show how the pressure unbalance increases when attempting to obtain more length of valve seat by using narrow width plates in greater number compared to fewer wide plates. Very important design compromise must be made in the valve and structural selection.

For the purpose of keeping the comparison between the structures of Fig. 1 and Fig. 2 as accurately as possible, the space for the accommodation of discharge valves will be within the cylinder bore.

Figs. 5, 6, 7, 8 and 9 are made to scale. The space between partition and plate edge and between plate edges is not necessarily the optimum space. It simply represents a minimum space chosen for the purposes of this discussion. From this type of sketch it is possible to calculate valve seat length using different widths of plates. Seat length is considered a very important item, because valve studies as well as practical experience has proven that there is certain maximum permissible valve lift to avoid damage by impact. Valves at the best are a compromise of a light weight (and perhaps lower strength structure) of minimum mass against a heavier structure of perhaps greater strength and greater mass, with a compromise of spring force to obtain a lesser resistance to opening or a spring force for a quick and positive closing.

The first examination for best valve selection will be that of the 3.7218 inch bore. Fig. 5 presents a sketch made to scale and in the top layout the one-half inch wide ring plate is used. Compare this with the lower sketch where the attempt is made to use the 5/16 inch wide ring plate. Structurally, it is possible to accommodate the two 1/2 inch wide plates but impossible to accommodate the three 5/16 inch wide plates. The radius of the center of the outer ring plate is:

$$\frac{3.7218 \text{ D}}{2} - (.125 + .250) = 1.4859 \text{ inches.}$$

The radius of the center of the inner ring plate is:

$$1.4859 \text{ R} - (.250 + .250 + .250) = .7359 \text{ inches.}$$

The total seat length for gas passage is:

$$1.4859 \times 2 \times 3.1416 \times 2 = 18.67 \text{ inches}$$

$$.7359 \times 2 \times 3.1416 \times 2 = 9.33 \text{ inches}$$

$$28.00 \text{ inches to be re-}$$

corded under column 1, item (m1) of Table 2.

The next examination is that of the 3.962 inch bore. Fig. 6 shows the sketch using the two 1/2 inch wide plates. The total seat length is determined exactly as above and totals 30.94 inches which is recorded under column 2, item (m1).

Fig. 7 is the 3.962 inch bore and using three 5/16 inch wide plates. The results show a questionable structure limitation at the center. There is, however, a total of 42.87 inches of valve seat length and that would make it advisable for the designer to make a more completely detailed structure of this valve assembly to determine if it is usable. This is recorded under column 2, item (m2).

Fig. 8 is the 5.640 inch bore and using three of the 1/2 inch wide plates it is possible to obtain 63.019 inches of valve seat.

Fig. 9 is also with the 5.640 inch bore but using 4 of the 5/16 inch wide plates. This provides 84.84 inches of valve seat.

All of these results are tabulated in item (m1) or (m2) of Table 2 under the column of the bore size applicable.

Item (i) of Table 2 records the cu. in. per second flow rate necessary to fill the displacement. Dividing this flow rate by the inches of valve seat, item (m), gives the cu. in. per second per inch of valve seat and these are recorded in item (n), 1 or 2 as applicable.

Item (c), Table 2, is the full displacement volume per cylinder and item (i) is the same volume in terms of a flow rate required to pass such a volume in the time of 1/2 crankshaft revolution. It is considered practical to use* such a flow rate for discharge as well as inlet comparisons because any interference to incoming vapors such as re-expansion by remaining vapor in the clearance area is sufficiently proportional to the decrease in time for vapor entrance and the remaining displacement to be filled that the flow rate remains practically the same. This is also true of discharge. A pressure increase before discharge reduces the volume to be discharged in about the same proportion as the reduction in time.

The next area of investigation is that of vapor inlet valve size and location. Much can be said about inlet valve area relative to discharge valve area and for the purposes of this discussion we will cause the inlet valve area to be no less than the discharge valve area and at the same time maintain the required minimum structure and seek a minimum of clearance volume.

Item (m1) under column 1 of Table 2 shows 28 inches of seat available for vapor passage in the selected discharge area. The radius of the center line of equal inlet valve seat will be:

$$\frac{28}{2} = R^2 \times 3.1416 \quad R = 2.11 \text{ inches}$$

The structure to accommodate this inlet valve in a compressor form such as Fig. 2 is illustrated in Fig. 10.

The computed radius of 2.11 inches is insufficient to permit the accommodation of the valve external of the space assigned to the discharge valves. In Fig. 10 the inlet valve has been illustrated at the least diameter possible, and with the least clearance space for vapor passage. This vapor passageway adds to clearance volume. The amount is shown on Fig. 10 and is recorded under column 1, item (k2).

Similar computations and sketches are shown by the sketches Fig. 11 and Fig. 12. The computed clearance volumes are recorded under columns 2, 3 and 4 respectively, items (k2).

A correct valve selection can be made only after a complete study of the pressure and temperature limits of the various refrigerants previously named. Time will permit a quick look at two of these refrigerants.

With refrigerant R-717 (ammonia) it is suggested that 100 degrees F be considered a top practical condensing temperature. Also, that a 10 compression ratio is the design goal. This means 211.9 PSIA condensing pressure and 21.19 PSIA inlet pressure. The vapor inlet temperature is the unknown and will vary greatly in field application; assuming it to be 30 degrees F, the expected discharge temperature plotted on the P.E. diagram will be approximately 345 degrees F.

With refrigerant R-22 it is suggested that 120 degrees F be considered the top practical condensing temperature. This means 274.8 PSIA condensing pressure and 27.48 PSIA inlet pressure at the 10 ratio and with an inlet vapor temperature of 30 degrees F, the expected discharge temperature will be approximately 238 degrees F.

Fig. 4 reminds us that the temperatures given above will actually be considerably different inside our compressor dependent on the valve selection. The 5/16 inch wide ring plate presents an internal overcompression which causes us to discard it from use in this design.

Fig. 13 was prepared for the purpose of illustrating the extent to which early designers went for the purpose of avoiding internal overcompression. Note that the seat of the valve button was formed so as to provide as near to a line contact as possible. For an example of the differences of overcompression, the safety head spring loading for the poppet type valve illustrated in Fig. 13 was the equivalent of 20 PSI per square inch of safety head exposure. Some present-day compressors with ring plate type valves have safety head spring loading equivalent of 170 PSI per square inch of safety head exposure.

The 1/2 inch width ring plate is selected for this design and the next consideration will be that of cylinder arrangement and choice of structure.

Fig. 14 is drawn to scale. It shows the most compact arrangement of cylinders, using four rods on a single crankshaft throw. The sketch is a composite of the Table 2, column 3 and column 4 compressor sizes because our choice will be made from one of these two. On the extreme left of the cylinder layout Fig. 14 the illustration is that of the Fig. 1 type at the longer stroke of the 900 RPM compressor. The next cylinder to the right is that of the same type with the shorter stroke of 1200 RPM. The next cylinder to the right of the vertical center is that of the Fig. 2 type at the longer stroke of 900 RPM and the furthest cylinder to the right is that of the Fig. 2 type at the shorter stroke of 1200 RPM.

The reason for the choice being between the column 3 and column 4 compressor is that of physical structure. The compressors under column 1 and 2 will require a 4 throw crankshaft with a center bearing and 16 cylinder bores as compared to half as many throws and cylinder bores for the 8 cylinder compressor.

Item (k3) in Table 2 is the volume of the recess space into the valve plate and that is assumed as 2.705 cu. in. Note that this is not an accurate volume because the valve assembly has not been fully completed. It is reasonably accurate for comparison purposes.

Item (k4) in Table 2 and recorded in columns 3 and 4 only is the total clearance volume if the Fig. 1 type compressor is selected.

Item (k5) in Table 2 and recorded in columns 3 and 4 only is the total clearance volume if the Fig. 2 type compressor is selected.

Item (L1) in Table 2 and recorded in columns 3 and 4 only is the total clearance volume of the Fig. 1 type divided by the cylinder volume.

Item (L2) in Table 2 and recorded in columns 3 and 4 only is the total clearance volume of the Fig. 2 type divided by the cylinder volume.

From a minimum clearance viewpoint:

The BEST DESIGN	FIG. 1 TYPE AT	900 RPM	.0392
2nd "	"	FIG. 1 TYPE AT	1200 RPM .0522
3rd "	"	FIG. 2 TYPE AT	900 RPM .0618
4th "	"	FIG. 2 TYPE AT	1200 RPM .0823

Other reasons for selection of the Fig. 1 Design are as follows:

- (a) Observe in the Fig. 1 structure the flow path of the incoming vapors. They first pass through a cylinder wall which is relatively cool because the water jacket is an excellent barrier to conductivity along the cylinder wall. The first contact of the incoming vapors with a hot surface is at the inlet valve in the piston head. In the Fig. 2 design, the inlet valve

structure to the left causes the inlet vapors to be in contact with the hot portion of the cylinder wall. Water cooling overhead does not have the direct cylinder wall cooling. The inlet valve structure to the right will have the greatest inlet vapor heating.

- (b) Inlet valves in the piston in the Fig. 1 type do not have any spring resistance to opening. The cylinder is filled with gas at more nearly the external pipe pressure than is possible with the Fig. 2 type where there is considerable spring resistance to opening.
- (c) The structure of the Fig. 1 type permits water jacketing to be most advantageously located. It provides the necessary barrier to heat flow from the discharge area into the inlet area.

All of the above account for the reason that Fig. 1 type compressors are in service at higher compression ratios than Fig. 2 type compressors.

Structurally, the Fig. 1 type permits closer cylinder centers than the Fig. 2 type.

A consideration of the effects of liquid refrigerant entering the compressor and the accumulation of oil under long periods of cylinder bypassing when unloaded has not been undertaken. They require serious consideration but time does not permit a discussion of design improvements which can aid in reducing possible hydraulic damage. An example: Fig. 3 is representative of a structure wherein hydraulic accumulation does not occur because there is continuous drainage into the discharge clearance area and gas movement discharges the hydraulic immediately.

In conclusion, the Author hopes that the "Guide Lines" suggested may be of assistance to those wishing to design better compressors for the ever-growing Refrigeration requirements.

TABLE 1 WORKING RATES CURRENT COMPRESSOR DESIGNS - SINGLE ACTING									
1	RPM	1800	1800	1200	1200	1200	900	720	327
2	RPS	30.00	30.00	20.00	20.00	20.00	15.00	12.00	5.45
3	Sec. Per Rev.	.0333	.0333	.0500	.0500	.0500	.0666	.0833	.1830
4	Sec. Per 1/2 Rev.	.01666	.01666	.02500	.02500	.02500	.03333	.04160	.09150
5	Bore Inches	3.2500	3.6875	3.7500	4.5000	4.9200	4.3750	6.7500	12.0000
6	Stroke Inches	2.7500	2.7500	3.0000	3.5000	2.9500	4.2500	5.0000	12.0000
7	Bore/Stroke	1.1750	1.3450	1.2500	1.2850	1.6680	1.0294	1.3500	1.0000
8	Av. P. S. Ft./Min.	825.00	825.00	600.00	700.00	590.00	637.50	600.00	654.00
9	Est. Min. H to P In.	.0050	.0050	.0055	.0065	.0055	.0080	.0095	.0200
10	% of Stroke	.182	.182	.183	.185	.186	.188	.190	.166
11	Cu. In. Per Cyl.	22.839	29.368	33.134	55.650	56.084	65.769	178.923	1357.170
12	Cu. Ft. Per Cyl.	.0132	.0170	.0191	.0322	.0324	.0380	.0135	.7854
13	Rate to Fill Cu. In./Sec.	1370.88	1762.78	1325.56	2226.00	2243.36	1973.26	4300.55	14832.45
14	Bore Cir. In.	10.210	11.585	11.781	14.137	15.456	13.744	21.206	37.699
15	Cu. In./Sec. Per In. Bore	134.26	152.25	112.51	157.45	145.15	143.55	202.79	393.43

TABLE 2 500 CFM DISPLACEMENT DESIGN					
		1	2	3	4
a	RPM	1800	1200	1200	900
b	CFM Per Rev.	.2777	.4166	.4166	.5555
b1	Cylinders	16	16	8	8
c	Cu. In./Cyl./Rev.	30.00	45.00	90.00	120.00
d	Piston Speed Ft./Min.	825	720	720	720
e	Bore Inches	3.7218	3.0960	5.6400	5.6400
f	Stroke Inches	2.750	3.600	3.600	4.800
g	Bore/Stroke Ratio	1.354	1.107	1.566	1.175
h	1/2 Rev. Seconds	.01666	.02500	.02500	.03333
i	Rate to Fill Cu./In./Sec.	1800	1800	3600	3600
j	Piston to Head In.	.008	.008	.008	.008
k1	Piston to Head Vol. Cu. In.	.866	.980	1.995	1.995
k2	Inlet Area Vol., Fig. 2 Design	1.850	2.055	2.705	2.705
k3	Recess Area Cu. In., Figs. 1 & 2 Design	--	--	2.750	2.750
k4	Total Vol. Fig. 1, Cu. In.	--	--	4.700	4.700
k5	Total Vol. Fig. 2, Cu. In.	--	--	7.405	7.405
L1	k4 / C	--	--	.0522	.0392
L2	k5 / C	--	--	.0823	.0618
m1	Est. Valve Seat In. - 1/2" Plate	28.00	30.94	63.919	63.919
m2	Est. Valve Seat In. - 5/16" Plate	--	42.87	84.84	84.84
n1	Cu. In./Sec./In./Seat-1/2" P.	64.28	58.20	56.35	56.35
n2	Cu. In./Sec./In./Seat-5/16" P.	--	41.90	42.43	42.43

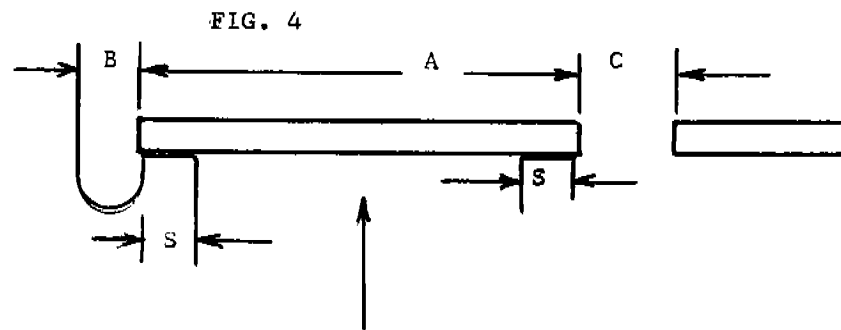
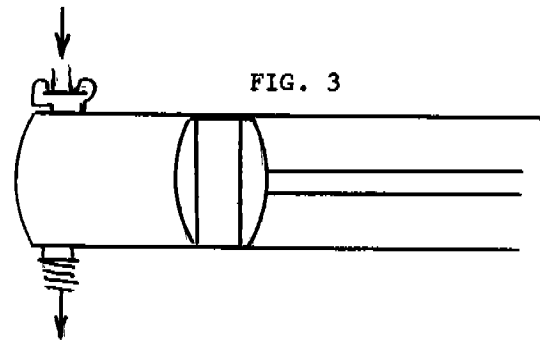
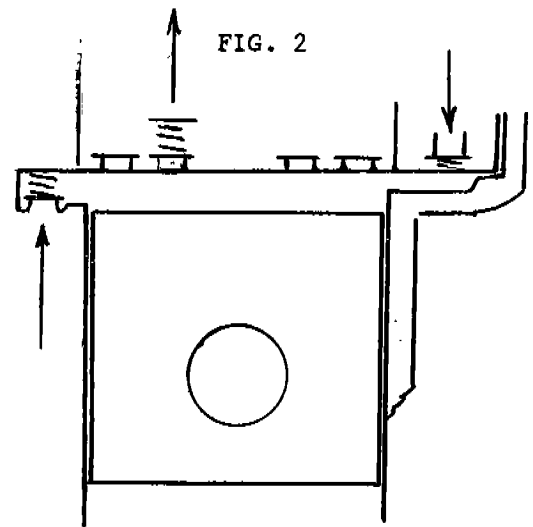
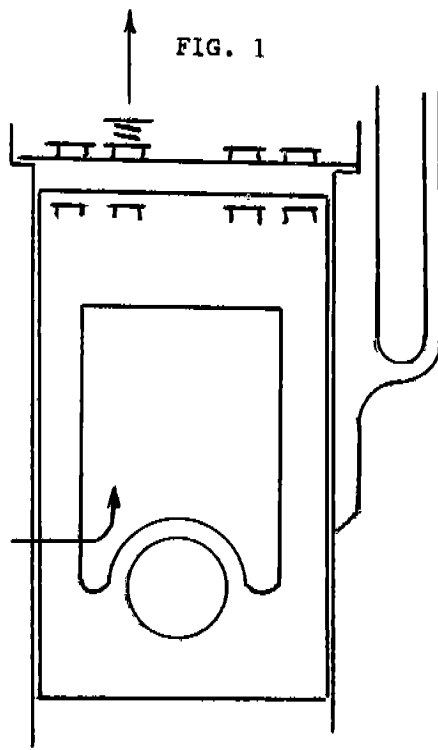


Plate Width Inches	Seat Width Inches	Inside Pres. Width	Inside Pres. Over Pres. At A	B Min. Inches	C Min. Inches	Center Dist. Inches
3/4 .750	1/16 .0625	5/8 .625	1.200	1/8 .125	1/4 .250	1.000
1/2 .500	1/16 .0625	3/8 .375	1.333	1/8 .125	1/4 .250	3/4 .750
5/16 .3125	3/64 .046875	7/32 .21875	1.428	1/8 .125	1/4 .250	9/16 .5625

